Gas Turbine Based Power Cycles - A State-of-the-Art Review

R. K. Bhargava*, M. Bianchi**, A. De Pascale**, G. Negri di Montenegro** and A. Peretto**
*Foster Wheeler USA Corp, 2020 Dairy Ashford Road, Houston, TX 77077, USA
**DIEM, University of Bologna, viale Risorgimento 2, 40136, Bologna, ITALY

Abstract: Gas turbines have been used in wide ranging applications since their world's first use in aviation and power generation in the jet engine powered flight of Heinkel aircraft (model He-178) and Brown Boveri & Cie's (BBC) 4 MW power generation plant in Neuchatel, Switzerland, respectively during 1939. This paper provides the historical evolution of the gas turbine (GT) based power cycles. A detailed parametric thermodynamic cycle analysis is presented for various GT cycles (mostly, which have been implemented). In addition, a comparative performance evaluation of various cycles is presented clearly showing ranges within which a particular arrangement can be beneficial. The simulation results are compared with the performance of existing machines with similar design conditions. A discussion is presented to show limitations and advantages of each GT cycle and the associated technological advancements made. To complete the review, modified Brayton cycles under development by the gas turbine manufacturers, researchers, etc. have also been identified.

Keywords: Gas turbines, Modified Brayton cycles, Thermodynamics, Technology review

1 INTRODUCTION

Fossil fuels (coal, oil and natural gas) are the major and depletable source of energy globally and 85% of the total energy consumed by the end of 20th century came from the fossil fuels [1]. This trend is not expected to change in the 21st century with the increasing demand on energy from some parts (such as China and India) of the world than the others. Recently, Bhargava [1] noted that more than one-third of energy for electric power generation globally came from coal in the last 2-3 decades. In the power generation market worldwide, gas turbines are becoming preferred choice because of their high cycle efficiency, small installation time, and low level of CO_2 emissions [2] and single digit NO_x emissions compared to the other power generating systems. In the coming years, advances in the gas turbine technologies combined with innovative modifications to the Brayton cycle, such as, Cascaded Humidified Advanced Turbine (CHAT) cycle, advanced Integrated Gasification Combined Cycle (IGCC), GT and high pressure solid oxide fuel cell (SOFC) hybrid cycle, and combination of high pressure SOFC with inter-cooling, Humid Air Turbine (HAT) cycle, etc. are expected to provide gas turbine based power plants with plant efficiency of 75% or more as envisioned by the U.S. Department of Energy (DOE) in their Vision-21 program initiated in 1999-2000 [3]. Gas turbine technology with fuel flexibility and comparatively lowest cost/kW will play an important role in coming decades in the power generation market.

This paper provides a brief summary on the evolution of gas turbine based power cycles including performance simulation of various cycles most of which have been implemented. Because of the space constraints, simulation results of the following gas turbine cycles are included in this paper: simple Brayton, Recuperated (REC) Brayton, Inter-Cooled Brayton (IC), Wet Inter-Cooled (IC Wet) Brayton, Inter-Cooled Recuperated (ICR) Brayton, Re-Heat (RH) Brayton, Inter-Cooled, Re-Heat, and Recuperated (ICRHR) Brayton, Cheng, and Below Ambient Gas Turbine (BAGT) cycles. The simulated performance is compared with the performance data of the commercially available gas turbines to identify advantages, limitations, and robustness of the simulation models used.

2 HISTORICAL BACKGROUND: EARLY GAS TURBINE DEVELOPMENT

John Barber was granted the world's first gas turbine patent, in 1791 in England, for his design that used the thermodynamic cycle of the modern gas turbine but obviously not the similar components. His design consisted of a chain driven reciprocating compressor, a combustion chamber and a turbine [4]. In 1903, Norwegian engineer Aegidius Elling successfully designed and built the first simple cycle gas turbine with a net power output of 8.1 kW and turbine inlet temperature (TIT) of 400°C (752°F) [5,6]. The low net turbine power output was attributed among other reasons to the fact that the gas turbine was designed to supply high pressure air. This gas turbine consisted of a 6 stage radial compressor and a single stage radial turbine. To accommodate limitations of the then existing turbine material strength, combustion gases were passed through a water/gas heat exchanger to cool gases and generate steam. Subsequently, a mixture of cooled combustion gases and steam were expanded in the turbine. Elling's thinking was much ahead of his time as is evident from the fact that to reduce the work of compression, he had incorporated the concept of intercooling and also implemented steam injection.

During the same time when Elling's work was in progress, Franz Stolze, a German engineer who got patent in 1899 for his gas turbine that was designed in 1873, was installing

world's first complete axial design GT at Berlin-Weissensee Power Station in Germany and tested it in 1905 [7]. This gas turbine, with net power output of approximately 150 kW had a single silo combustor, a recuperator, and more stages in the turbine (15 axial stages) than in the compressor (10 axial stages) (See also, Table A1 in Appendix A). It must be noted that the silo design combustors have been used by ABB (now Alstom Power) in many gas turbines with large power rating such as the one installed at 300 MW compressed air plant at Huntorf in Germany in 1977 consisting of two silo combustors [8].

Another gas turbine of historical importance, developed by Charles Lamale and Rene Armengaud and designed by French engineer August Rateau in 1905-1906, consisted of a BBC supplied 25 stage centrifugal compressor in three casings with intercooling and a two stage impulse design axial turbine [4]. The unit, designed to produce 400 hp with TIT of 400°C and the cycle pressure ratio (β) of 3 had a cycle efficiency of only 2%~3% because the compressor consumed 328 hp.

One of the problems with the early developments in the gas turbine technology was associated with the fact that the compressor section consumed significant amount of power produced in the turbine section including limited understanding of the aerodynamics of compressors and turbines.

In 1920's and 1930's considerable amount of efforts were made at BBC (later ABB and now Alstom Power) in developing efficient axial compressors. The work of Ludweig Prandtle in Germany, one of the world's pioneers in the field of aerodynamics, on the development of thin and cambered airfoils theories combined with the development of expanding single airfoil theory to the multistage axial compressor by C. Seippel at BBC played a significant role in enhancing the axial compressor technology [8].

The world's first successful electric power generating gas turbine consisting of a single shaft design with a 23 stage axial compressor, one single can combustor and a seven stage axial turbine, developed by BBC went into commercial operation in Neuchatel, Switzerland in 1939 [9]. This gas turbine with TIT of 550°C (1022°F), β of 4.4, and power generation capacity of 4000 kW at the generator terminal had an overall cycle efficiency of 17.4% [8,9]. The axial compressor consumed approximately 74% of the power produced by the turbine. Compressor and turbine had an adiabatic efficiency of 85% and 88% respectively. On September 2, 1988, ASME (American Society of Mechanical Engineers) declared the gas turbine at Neuchtal as the international historic mechanical engineering landmark [9].

A performance summary of some of the early gas turbine systems utilizing various design changes and cycle modifications up to mid-twentieth century is given in Table A1 (See Appendix A). As is evident from Table A1, within a short span of 7 years after installation of the first successful commercial gas turbine at Neuchatel, the cycle efficiency was almost doubled even at a low value of TIT (see gas turbine System No. 7 in Table A1). This was realized by incorporating a complex cycle design with multiple intercoolers, reheat and recuperation.

It must be mentioned that some of the complex cycle designs could not be continued because of the availability of cheaper fuels in later years and high equipment cost (cost/kW) of modified Brayton cycles, and phenomenal technological advancements in aero-engines which were implemented in industrial gas turbines.

3 GAS TURBINE DEVELOPMENT IN THE SECOND HALF OF THE 20TH CENTURY

Until late 1950's, high temperature materials then available could allow TIT of 732°C (1350°F). 1960's onwards, considerable efforts were devoted to increase TIT for improving the Brayton cycle efficiency. For example, in 1961, Westinghouse commissioned their W191 gas turbine with power output of 18 MW at TIT of 788°C (1450°F) and cycle efficiency of 25.4% [10]. There was also an aggressive push for the development of combined cycle systems in 1960s to improve the overall efficiency of the gas turbine based power plants. The combined cycle with Brayton cycle as the topping cycle and Rankine cycle as the bottoming cycle was found a good fit that allowed the addition of heat energy at high temperature in the gas turbine and rejection of heat at low temperature in the steam turbine. Westinghouse placed in operation, at San Angelo's Power Station in the U.S. in 1967, their W301 gas turbine rated at 25 MW with TIT of 788°C (1450°F) and a combined cycle efficiency of 39% which remained highest in the U.S. for a number of years [10]. The effects of TIT and cycle pressure ratio on combined cycle performance clearly showed, as is evident from the available literature [10-13], that a GT with low value of β provides higher specific work and optimum cycle efficiency for the combined cycle systems. Since many mid to large size gas turbines were developed for the combined cycle applications, many manufacturers have optimized their gas turbines for the maximum specific work than for the maximum cycle efficiency. This approach to optimize the GT for maximum specific work is quite evident in Fig. 1 which provides trends of cycle pressure ratio and TIT for the commercially available selected gas turbines from the major GT manufacturers. Solid lines in Fig.1 correspond to values of β and TIT to achieve maximum cycle efficiency (η_{el}) and specific work (w_{el}). Also, for a given value of TIT, β value to achieve maximum cycle efficiency is higher than for the maximum specific work. The continuous increase in TIT value required implementation of cooling technologies for hot gas path components in addition to the development in materials and manufacturing technologies. Many blade cooling technologies, developed for aircraft engines, were transferred for their use in the industrial gas turbines included impingement, film and convective cooling.

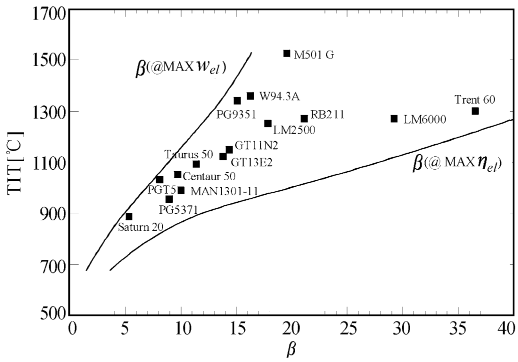

Fig.1 Cycle pressure ratio (β) and TIT for commercially available gas turbines with Brayton cycle

In mid-1980's, the major gas turbine manufacturers started developing F-class industrial gas turbines with TIT of 1260°C or higher which were commercialized in the early 1990's. This development work resulted in machines, the performances of some are summarized in Table A2 (see systems No. 4, 6-8). The first F-class machine (GE 7F) was installed by GE at Virginia Power's Chesterfield No. 7 Station in 1990 and had an efficiency of 45.2 at TIT of 2300°F with a total power output of 214 MW in the combined cycle mode (Note, in a simple cycle mode, this unit was rated at 150 MW and 34.5% efficiency) [14]. Another machine, model 501F, jointly developed by Westinghouse, Mitsubishi Heavy Industries and Fiat Avio, and which became operational in 1993, with TIT of 1260°C (2300°F) and power output of 160 MW, had cycle efficiency of 35.6% and 51.7%, in simple and combined cycle applications, respectively [10].

In the early 1990's, the U.S. DOE in partnership with major gas turbine manufacturers, universities, and national laboratories initiated a program to develop the Advance Turbine Systems (ATS). One of the main objectives was to develop advance turbine systems (industry and utility size gas turbines) with combined cycle efficiency of 60% [15-19]. ATS program resulted in the introduction of high efficiency machines (see systems 9-13 in Table A2).

In 1999-2000, Vision-21 program was initiated by DOE to develop fuel flexible electric generation facilities with the following main objectives: (a) coal based systems to achieve 60% efficiency; (b) natural gas based systems to achieve 75% efficiency; and (c) near zero emissions of sulfur, NO_x, particulate matters, and 100% reduction of CO_2 with sequestration [3]. This program aims to introduce advanced technologies by 2015.

Since the early historical development in the gas turbine technology, engineers and developers have always strived to achieve gas turbines with higher cycle efficiency. As a result, numerous technological advancements including but not limited to such as the development of high strength materials, high temperature coatings, blade cooling methods, blade designs and cycle modifications have been made.

There are other numerous modified Brayton cycles which have been examined, some as a part of the Vision-21 program, whereas, others by independent researchers and developers which might become commercially available in the future. These cycles include, advanced Integrated Gasification Combined Cycle (IGCC) [20], Evaporative Gas Turbine (EGT) cycle [21], Humid Air Turbine (HAT) cycle [22], Cascaded Humidified Advanced Turbine (CHAT) cycle [23], high pressure Solid Oxide Fuel Cell (SOFC) and GT hybrid cycle [24], high pressure SOFC and Inter-Cooled (IC) hybrid cycle [25], high pressure SOFC and HAT hybrid cycle [25], and solar-hybrid GT cycle [26].

Investigations and development of prototype of some of the aforesaid hybrid gas turbine systems have shown promising results [20-26]. For example, preliminary studies by Rao et al. [25] showed that hybrid systems of IC GT with high pressure SOFC and SOFC with HAT cycle could provide cycle efficiency of 75% or more. Another interesting hybrid system, consisting of combining GT with the freely available solar energy and recently reported by Schwarzbozl et al. [26], could achieve conversion efficiency from the solar heat up to 50%. More research and development works are ongoing to achieve the goals set by the Vision-21 program. It must be mentioned that performance simulation of these cycles are not covered in the present study because of the space limitation.

4 PARAMETRIC PERFORMANCE ANALYSIS PROCEDURES AND ASSUMPTIONS

For the selected GT cycles as identified in the Introduction, a parametric performance analysis has been carried out which can help in identifying suitable design parameters for each cycle. The analysis has been performed for five different GT technology levels, defined in terms of TIT and polytropic efficiency values (see Table 1). The parametric study was aimed at covering the range of values representing machines used during early phase of GT development and machines being currently used and or developed for future applications. In particular, the polytropic efficiency values given in Table 1 have been chosen following the work of Consonni and Macchi [27] that provides a correlation between the TIT value and the engine size.

The effect of hot gas path components cooling has been included in the study for all the levels of considered TIT, except for the case of TIT = 700°C. In particular, in the case of TIT = 1500°C the turbine includes three cooled stages, in case of TIT = 1300°C two stages are cooled and only one stage is cooled for the case of TIT = 1100°C and TIT = 900°C. For the particular cases of RH, ICRH, and ICRHR GT cycles, the one-stage high pressure turbine (HPT) was assumed to be cooled for all TIT values except 700°C technology level.

Table 1 Different technologies investigated

GT Tech. Level	Firing Temp. (TIT) (℃)	No. of Cooled Stages*	Compressor Polytropic. Eff.	Turbine Polytropic. Eff.
A	1500	3	0.91	0.89
B	1300	2	0.90	0.89
C	1100	1	0.87	0.88
D	900	1	0.81	0.87
E	700	-	0.74	0.85

* Plus HPT single stage, in case of RH, ICRH and ICRHR

Table 2 Assumptions common to all cycles

Parameter	Value	Unit
Fuel LHV	47500	kJ/kg
Combustion efficiency	0.99	-
Generator + mech. loss efficiency	0.98	-
Inlet pressure losses	1	kPa
Outlet pressure losses	1	%
Pressure drop through combustors	3	%

Table 3 Specific assumptions for analyzed GT cycles

Cycle	Parameter	Value	Cycle	Parameter	Value
REC	ε_{REC}	90%	RH	β_{HPT}	2
	Δp_{REC}	3%		$m_{\text{cool 2CC}}$	6% of inlet air
IC	β_{LPC}	2.5	ICRH	β_{LPC}	2.5
	ε_{IC}	90%		ε_{IC}	90%
	Δp_{IC}	3%		Δp_{REC}	3%
	T_{IC}	40°C		T_{IC}	40°C
	$T_{\text{water IC}}$	30°C		β_{HPT}	2
Wet IC	β_{LPC}	2.5		$m_{\text{cool 2CC}}$	6% of inlet air
	$T_{\text{water inj}}$	15°C	ICRHR	β_{LPC}	2.5
	T_{IC}	40°C		ε_{IC}	90%
ICR	β_{LPC}	2.5		Δp_{IC}	3%
	Δp_{IC}	3%		T_{IC}	40°C
	ε_{IC}	90%		β_{HPT}	2
	T_{IC}	40°C		$m_{\text{cool 2CC}}$	6% of inlet air
	ε_{REC}	90%		ε_{REC}	90%
	Δp_{REC}	3%		Δp_{REC}	3%
STIG	$m_{\text{steam inj}}$	7% of inlet air	BAGT	p_{low}	0.4 bar
	$T_{\text{steam inj}}$	350°C		ε_{IC}	90%
	$p_{\text{steam inj}}$	130% of $p_{IN\,CC}$		Δp_{IC}	3%

The cooling air mass flow rates for rotor blades and nozzles of each cooled stage are calculated on the basis of the stage inlet hot gas and coolant air temperature following the work of El-Masri [28], assuming film cooled blades. For the case of highest TIT value considered, in order to calculate the stage inlet temperature, the stage loading on the first three cooled stages was selected to be 260 kJ/kg, 250 kJ/kg and 220 kJ/kg, respectively. It must be noted that these assumed values of stage loading are within the range of exiting gas turbine stages. In the case of TIT of 1300°C, the first two cooled stages are assumed to have stage loadings of 260 kJ/kg and 250 kJ/kg, respectively. Whereas, for TIT values of 1100°C and 900°C the cooled stage is considered with stage loading of 260 kJ/kg.

The other common and specific assumptions adopted for simulations of all the analyzed cycles are summarized in Table 2 and Table 3. In particular, for the inter-cooling concept, the value of temperature achieved after cooling, T_{IC}, is assumed 40°C, and the inlet cooling water temperature, $T_{\text{water IC}}$, is considered to be 30°C, conditions compatible with a dry-cooling tower. For the wet IC cycle, the injected water temperature, $T_{\text{water inj}}$, is assumed 15°C. The LPC (Low Pressure Compressor) pressure ratio of IC cycles (β_{LPC}) is chosen equal to 2.5, close to the conditions of maximum cycle efficiency, following the work of Bettocchi et al [29].

The HPT pressure ratio is chosen equal to 2, sufficiently low to maximize the specific work output [30]. The second combustor chamber for the cycles adopting reheating has been considered cooled, with a fixed amount of air mass flow ($m_{cool2CC}$) equal to 6% of the total GT inlet air.

The turbine discharge pressure (p_{low}) in the BAGT cycle simulations has been assumed 0.4 bar corresponding to the maximum cycle efficiency as shown by Bianchi et al. [31].

All the cycles have been considered operating at ISO conditions (ambient temperature: 15°C, pressure: 1.013 bar, relative humidity: 60%). The performance analysis has been conducted using a commercial computer code (GateCycle, version 5.22) taking into account the actual composition and the specific heat variation with temperature of the gas evolving in the cycle.

5 PERFORMANCE OF SIMULATED GT CYCLES–RESULTS AND DISCUSSION

In this section, results obtained from the simulation of selected cycles most of which have been adopted in the industry are presented and discussed. Practical implications of each evaluated cycle are also discussed.

Brayton Cycle

The simple Brayton cycle is evaluated first, considering the assumptions and calculation procedures outlined earlier, so that the performance of the other modified Brayton cycles can be compared with respect to the reference cycle.

This GT cycle has been extensively evaluated over the years and its performance trends are well known: (1) the cycle efficiency increases with increase in the value of TIT and β; (2) for a given TIT, there are optimum values of β

corresponding to which the cycle can have maximum efficiency and maximum specific work; and (3) as shown earlier in Fig.1, the optimum value of β is higher for maximum cycle efficiency compared to its value for maximizing the specific work.

The presented results in Fig.2 show that at TIT = 1300℃, maximum efficiency (42.3%) could be achieved at a pressure ratio of 38. Whereas, optimum cycle pressure ratio to maximize specific work is 14 at TIT of 1300℃. In comparison, the existing gas turbine, Rolls-Royce Trent 60 with TIT = 1288℃, β = 35, has a cycle efficiency of 41.3% at ISO conditions. The comparison should be considered with caution as our calculations do not simulate any specific machine and many assumptions have been made.

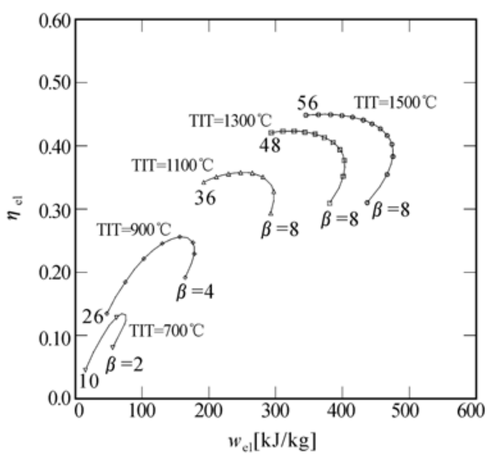

Fig. 2 Cycle efficiency versus specific work – Brayton Cycle

Recuperated (REC) Brayton Cycle

In the recuperated Brayton cycle, heat energy is recovered from the gas turbine exhaust and used to preheat the combustion air. The recovery of waste heat from the exhaust gases reduces the amount of fuel required to achieve the desired TIT. This approach has been implemented extensively in 50's and 60's to improve the GT efficiency. Numerous gas turbines used in pipeline applications utilized this approach to boost GT efficiency. In 1980's the world's first gas turbine powered battle tank, the US Army main battle tank M1 Abrams, was powered by a 1500 hp Lycoming AGT 1500, a three-shaft recuperated engine [32]. In the early development phase of the gas turbines, recuperated gas turbines typically achieved efficiencies of 21%~23% as compared to the then achievable simple cycle efficiency of 15%~18% [33]. Since the recuperator utilizes the temperature difference between the exhaust gases and the compressor discharge, efficiency gain compared to the Brayton cycle is relatively higher at low β value and high TIT.

One of the limitations of using recuperation is that in order to recover heat via the recuperator, the turbine exit gas temperature must be higher than the compressor exit temperature. This constraint allows to adopt regeneration only in the case of GT engines with low β values. Moreover, the recuperator heat exchanger introduces a pressure drop that reduces the cycle thermodynamic efficiency. Another aspect that has to be considered in adopting regeneration is the additional cost of the recuperator.

As evident from Fig.3, efficiency improvement of 6%~7% point could be achieved at TIT of 1100°C requiring specific work of 250 kJ/kg. Also, to achieve specific work of 250 kJ/kg, REC Brayton cycle requires a pressure ratio of less than 6 compared to 24 for the simple cycle.

Another recuperation technique evaluated and discussed in the literature [34-37] involves chemical recuperation, a cycle termed chemically recuperated gas turbine (CRGT) cycle, where exhaust heat is recovered in a Heat Recovery Steam Generator (HRSG) in which superheater section is replaced by a Methane Steam Reformer (MSR). The natural gas is mixed with the steam and fed in the MSR. In the MSR, the mixture of steam and fuel is heated by the GT exhaust gases and endothermic reaction occurs between fuel and steam, resulting in a larger energy recovery compared to the conventional recuperator. One of the main advantage of CRGT cycle is its potential for ultra-low NO_x emissions due to the presence of steam in the reformate gas.

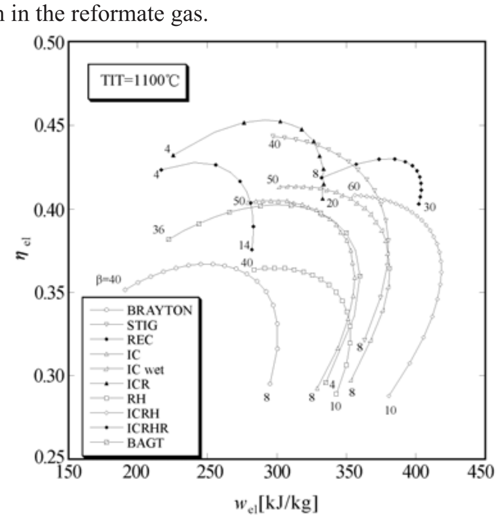

Fig.3 Comparative performance of investigated cycles – Cycles efficiency versus specific work, TIT=1100℃

Inter-Cooled (IC) Brayton Cycle

The compressor section of the gas turbine consumes more than two-third of the power produced by the gas turbine. Some of the earlier machines consumed more than 70% [8] of the total gas turbine power. It is quite evident from Fig. 4 that the gas turbines with TIT of 1200°C and higher (developed after 1980's) have lower values of compression work ratio (ratio of compression to turbine work). This improvement can be attributed to the advancements made in the compressor technology and cooling methods for the hot gas path components. Compression work ratio values, presented in Fig. 4, have been estimated corresponding to the ISO ambient conditions for a large number of commercially available gas turbines.

One of the approaches used to reduce the compression work has been to use intercooler between one or more sections of the compressor. The intercooling process can be

accomplished by a conventional heat-exchanger or by means of a water spray inside the gas stream (as accomplished in the SPRINT system by GE in LM 6000).

The IC cycle solution provides higher specific power compared to the simple Brayton cycle case operating with the same values of β, TIT and polytropic efficiency for expansion and compression; in particular, the intercooling process can be optimized in order to minimize the overall compression work. The cycle efficiency of the system does not increase compared to the simple Brayton cycle, because the compressor discharge temperature decreases in presence of the inter-cooling and thus more fuel is required for reaching the same TIT level. Moreover, the intercooling process requires a low temperature fluid source and the cold flow circuit increases auxiliary power requirements.

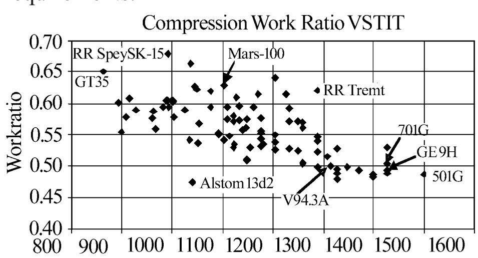

Fig.4 Compression work ratio vs turbine inlet temperature for commercially available gas turbines [38]

The Delaval Ljungstrom Turbine Co. (later ABB and now Alstom Power) built number of intercooled units in 50's and installed their first unit at Vastervik Power Station of the Swedish State Power Board in 1959 with cycle efficiency of 26% [33]. More recently in 2006, the world's most efficient intercooled gas turbine, GE's LMS-100 with cycle efficiency of 46%, TIT=1370℃, and β=42 went into commercial operation [39]. Our parametric study shows that at TIT=1300℃ and β=42, the cycle efficiency of 44.7% could be achieved a comparative performance prediction compared to the LMS-100 realizing difference in the TIT values (see Fig. 5).

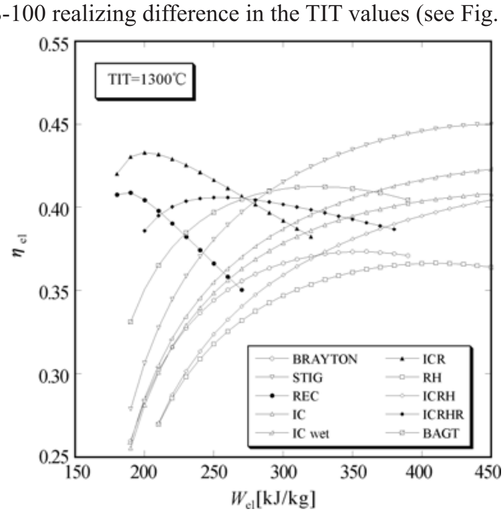

Fig.5 Comparative performance of investigated cycles-efficiency V.S. pressure ratio at TIT=1300℃

In principle, the IC cycle can be easily implemented, with the adoption of an intercooler heat-exchanger, on an aero-derivative engine originally equipped with a LPC (Low Pressure Compressor) and a HPC (High Pressure Compressor). As will be discussed later in the paper, the use of intercooling combined with recuperation improves performance of the Brayton cycle more than the non-intercooled recuperated cycle. The main reason being that compression work is reduced and the energy lost during the intercooling is recovered by additional energy recovery in the recuperator.

Another approach to implement intercooling effect is to inject liquid water into the gas stream between compressor sections (LPC and HPC sections) in a gas turbine similar to the approach adopted in GE LM6000 Sprint gas turbine. The injected water evaporates during compression inside the HPC, producing a decrease in the air temperature, resulting in the increased air mass flow rate and thus power augmentation per unit of inlet air mass flow. Our simulation shows that approximately 34% increase in the specific work and 2.9% point improvement in the cycle efficiency could be obtained at β of 28 and TIT of 1300°C compared to the simple Brayton Cycle at ISO conditions (see Fig.5 and Fig.6). For LM 6000 Sprint system, increase in power output and efficiency of 20% and 3.9% point, respectively at 32°C day has been reported by Jones and Jacobs [40]. A large gain in power output in our simulation can be attributed to the high ambient temperature used by Jones and Jacobs.

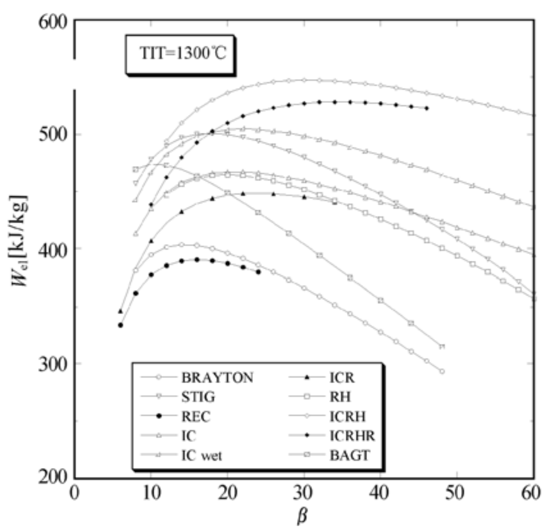

Fig.6 Comparative performance of investigated cyclesspecific work V.S. pressure ratio at TIT=1300℃

Inter-Cooled Recuperated (ICR) Brayton Cycle

This cycle configuration represents a solution to increase both specific power and efficiency of the simple Brayton cycle. As shown in Table A1, BBC installed a 10 MW unit in Romania with cycle efficiency of 23% at TIT=577℃ and β=12. In the United States in 1950, GE installed an ICR gas turbine at the Bangor Hydroelectric company with cycle efficiency of 28% at TIT of 815℃ and β of 9. The poor cycle efficiency of the ICR systems in early years can be mainly attributed to the limitation of materials used (low value of

TIT) combined with poor efficiency of GT components. In contrast, the latest ICR unit WR-21 developed by Westinghouse & Rolls-Royce and tested in 1997 provided 42% efficiency [41]. This machine derived from the Rolls-Royce RB211 and Trent aircraft engines has shown low specific fuel consumption under varying load conditions, an important requirement for the mechanical drive applications.

The main drawback of the ICR cycle is associated with two heat exchangers, leading to increased equipment costs and pressure drops. A study by El-Masri [42] showed higher cycle efficiency for the evaporative water injected intercooler compared to the intercooler of surface heat exchanger design. A comparative performance at TIT of 1100 ℃ shows improvement of approximately 3% points in efficiency for ICR compared to REC Brayton cycle (see Fig.3). This estimate is found in agreement with the work of El-Masri [42]. Furthermore, the optimum β values for both REC and ICR cycles are 10 or less (see Fig. 5) at 1300℃ with the assumed parameters in this study suggesting that this modification will be more suitable for small to mid-size gas turbines.

Re-Heat (RH) Brayton Cycle

The reheat cycle, also known as sequential combustion, takes advantage of the basic thermodynamic fact that the constant pressure lines diverge on a T-s diagram. As a result, work extracted in more than one expansion process will be higher than in a single expansion process between the two same pressure levels. The net work output of the cycle depends on the intermediate pressure of expansion. Heat is added in the second combustor until the limiting temperature is reached.

The reheat cycle is characterized by an increase of specific power output in comparison with a simple cycle operating with the same values of β, same first turbine TIT, and polytropic efficiencies of compression and expansion processes. For this reason, the RH cycle has been commonly implemented in military aero-engines to increase the thrust-to-weight ratio.

One of the benefits of using reheating is the availability of hotter gas at the engine discharge, in comparison with a simple cycle GT operated with same TIT level and β; thus, the hot gas at the exit can be employed in a HRSG of a combined cycle plant to produce superheated steam with higher enthalpy content.

BBC developed reheat cycle as early as in 1948 and later implemented in a 300 MW underground compressed air storage plant in Germany [8]. In 1994, a gas turbine with reheat, developed by ABB, became operational with cycle efficiency of 37.5% at TIT of 1260°C and β of 32 (see Table A2). This high cycle efficiency could be achieved with the use of Controlled Diffusion Airfoils in the compressor giving improved compressor performance under part-load operation and increased surge margin in addition to the effect of reheating [8].

The parametric performance analysis of the present study shows that a considerable amount (approximately 25%) of specific work can be increased by implementing reheat to the Brayton cycle at the same value of β and TIT (see Fig.6).

Inter-Cooled Re-Heat (ICRH) Brayton Cycle

The adoption of a complex ICRH cycle leads to an increase of the specific power output and, if the intermediate compression and expansion pressure levels are optimized, the overall efficiency is improved with respect to the simple cycle with same max pressure and temperature. As evident from Fig.6, this cycle could provide the highest amount of specific work over a wide range of β compared to all the other examined cycles at TIT = 1300℃. Also, this cycle has potential to achieve higher cycle efficiency compared to the RH cycle (see Fig.5). Whereas, higher value of cycle efficiency could be achieved compared to the Brayton cycle for $\beta > 35$.

Inter-Cooled, Re-Heat, and Recuperated (ICRHR) Brayton Cycle

A further increase in efficiency for an ICRH cycle can be obtained by including a recuperator in the system, obtaining the ICRHR cycle. For the ICRRH cycle, cycle performance is influenced by the following variables: pressure ratio (β), TIT, LPC pressure ratio (β_{LPC}), intercooler exit temperature (T_{IC}), reheat pressure, reheat exit temperature, intercooler effectiveness (ε_{IC}) and recuperator effectiveness (ε_{REC}). It must be noted that recuperation is necessary with various cycles such as IC, RH, and ICRH as these cycles are associated with the decrease in average temperature of the heat supply and increase in the average temperature of heat rejection. Subsequently, without recuperation the aforementioned cycles will have lower efficiency compared to when recuperation is used.

El-Masri [43] points out two important losses in a gas turbine cycle: (1) the first one is exergy destruction associated with the combustion process. This loss decreases as the mean combustion temperature is increased either by increasing the value of TIT or adding another combustion process (such as reheat); (2) the second source of exergy destruction is associated with the hot gas path component cooling related which will accompany with increase in TIT or with the implementation of reheat increase.

A commercial unit using such a complex Brayton cycle with power rating of 27 MW, TIT=650°C and β=8.1 and having cycle efficiency of 34% was installed by BBC at AG Beznau Station in Switzerland in 1946 (see Table A1). In the U. S., Dresser-Rand used ICRHR cycle gas turbine, rated at 110 MW, in a Compressed air energy storage application at the Alabama Electric Cooperative's plant in McIntosh which went into commercial operation in May 1991 [33,44].

Our simulation shows that ICRHR cycle over the range of pressure ratios has higher performance compared to the simple Brayton cycle, however its efficiency is lower than ICR cycle. Furthermore, this cycle does have higher specific work compared to the other cycles except ICRH cycle (see Fig.6). As expected, the recuperated ICRH cycle has higher cycle efficiency compared to the ICRH cycle (see Fig.5).

Cheng / STIG Cycle

Steam injected gas turbine cycle consists of a combined Brayton and Rankine cycles, without the need of a steam turbine and associated accessories, in which exhaust gas generated steam is introduced upstream of the turbine section. Injection of steam in the gas turbine combustor is a matured technology which is in use for NO_x emissions control. For power and efficiency enhancement of the gas turbine, steam can be injected in various locations: upstream of the combustor mixing with the primary and secondary air, through the fuel nozzles premixed with the fuel, ahead of the low pressure turbine, and ahead of the power turbine. The most significant contribution to the increased power output and efficiency is attributed to the extra mass flow through the turbine without the compression work. This cycle is commonly known as Cheng cycle named after D. Y Cheng who patented the concept in 1978 [45,46]. GE investigated the merits of steam injection in mid-70's and termed it STIG cycle.

The first commercial unit employing the Cheng cycle was Allison 501KB gas turbine which was installed at the San Jose University campus, California, USA, in 1984 to supply electricity and steam for power, heating, and cooling [46]. The estimated performance of 501KB with STIG was 70% increased output and 11% point (28% to 39%) efficiency improvement. Later in 1986, Cheng cycle was retrofitted on an operational GE LM5000 at the Simpson's Paper Co. in California, USA. Testing of the unit showed power increase from 34 MW to 49 MW and efficiency increase from 37% to 41% [47]. In addition to power and efficiency enhancement, Cheng cycle improves part-load performance.

Our parametric analysis shows a significant improvement in the values of specific work (25%) and efficiency (3.4% point) compared to the simple Brayton cycle at TIT=1300°C and β=18 (see Fig. 5 and Fig.6). Another variance to the STIG cycle has been studied consisting of LM5000 gas turbine implemented with STIG and intercooling and is termed ISTIG. LM5000 with ISTIG cycle is estimated to produce 110 MW power output at 47%~48% efficiency [45].

Below Ambient Gas Turbine (BAGT) Cycle

The Below Ambient Gas Turbine (BAGT) cycle involves discharging gas from the Brayton cycle at a pressure value below ambient pressure. Subsequently, exhaust gas is cooled in a heat exchanger and recompressed to ambient pressure level. Significant improvement in performance (both cycle efficiency and specific work) could be achieved compared to the simple Brayton cycle (see Fig.5 and Fig.6).

A comparison of optimum pressure ratio corresponding to the maximum values of cycle efficiency and specific work for different technology levels (TIT) reveals the following: (1) among the cycles examined, REC cycle requires the lowest value of β to maximize the cycle efficiency at a given value of TIT (see Fig.7a); (2) for a given value of TIT, BAGT cycle requires the lowest β value to maximize specific work (see Fig.7b). Whereas, for all other examined cycles, β value required to optimize specific work is larger than the optimum value of β required for a simple Brayton cycle; (3) the values of β corresponding to maximum cycle efficiency and specific work for each cycle increase with the increase in the TIT value.

Some additional observations based on the comparative performance of various cycles examined in the presented study reveal the following:

- For mid-level technology (TIT=1100℃) the ICR cycle provides the largest improvement of efficiency with low cycle pressure ratios (β=4-10). Whereas, for the best available technology (TIT=1500℃) the STIG cycle would allow to obtain peak efficiency values of 50% or more with β values approaching in the range of 60.
- The maximum amount of specific work could be achieved using the ICRH cycle with feasible values of β (near 20 at TIT=1100℃ and near 35 at TIT=1500℃).
- For the investigated range of TIT values, the recuperated cycles require lower values of β to maximize cycle efficiency compared to the non-recuperated cycles. For all values of TIT, addition of recuperation to the non-recuperated cycle reduced specific work as expected because of the recovery of waste heat.
- Among intercooled cycles, wet intercooling approach provides higher value of specific work at low value of β compared to the other intercooled cycles. The main reason being higher value of heat capacity of the working fluid. Also, comparatively equipment cost will be lower.

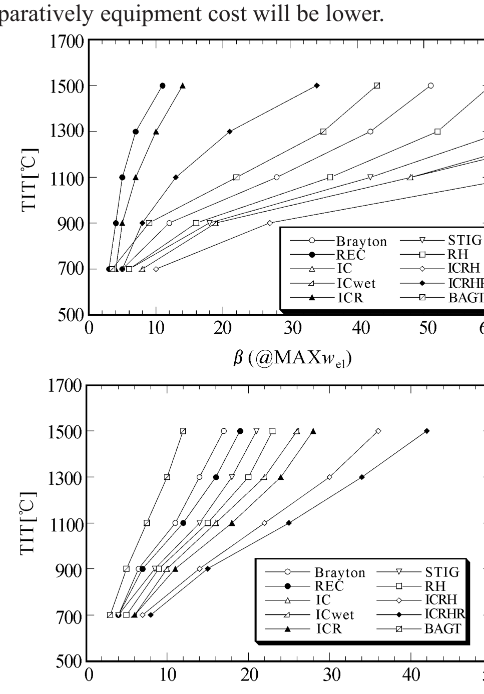

Fig.7 Comparative performances of investigated cycles –
(a) Optimum pressure maximizing cycle efficiency (top)
(b) Optimum pressure maximizing specific work (bottom)

6 CONCLUDING REMARKS

This paper has provided in brief a historical evolution of the gas turbine cycles since development of the first industrial gas turbine producing net work output in 1903. The results presented clearly show a reasonable comparison between the overall performances of a given cycle in comparison to the performance of commercially available gas turbines utilizing a similar cycle and design conditions. It must be noted that the economic evaluation is a very important factor when considering a particular cycle, which has not been considered in this study.

7 NOMENCLATURE

Acronyms

BAGT	Below Ambient Gas Turbine
CHAT	Cascaded Humidified Advanced Turbine
CRGT	Chemically Recuperated Gas Turbine
EGT	Evaporative Gas Turbine
EFGT	External Fired Gas Turbine
HAT	Humid Air Turbine
HPC	High Pressure Compressor
HPT	High Pressure Turbine
HRSG	Heat Recovery Steam Generator
IC	Inter-Cooled
ICR	Inter-Cooled Recuperated
ICRH	Inter-Cooled Re-Heat
ICRHR	Inter-Cooled Re-Heat Recuperated
IGCC	Integrated Gasification Combined Cycle
ISTIG	Inter-cooled STeam Injected Gas turbine
LHV	Lower Heating Value
LPC	Low Pressure Compressor
MSR	Methane Steam Reformer
REC	RECuperated/RECuperator
RH	Re-Heat
SOFC	Solid Oxide Fuel Cell
STIG	STeam Injected Gas turbine
TIT	Turbine Inlet Temperature

Symbols

$m_{\text{cool 2CC}}$	2nd combustor cooling mass flow rate
$m_{\text{steam inj}}$	injected steam mass flow rate
p_{low}	below ambient pressure
$p_{\text{steam inj}}$	injected steam pressure
T_{IC}	air temperature at IC exit
$T_{\text{water IC}}$	water inlet temperature to the IC
$T_{\text{water inj}}$	injected water temperature
w_{el}	specific work

Greeks

β	pressure ratio
Δp	pressure loss
ε	heat exchanger effectiveness
η_{el}	electric efficiency

REFERENCES

[1] R. K. Bhargava, "Global energy market – Past, present, and future," *ASME Paper* No. GT2006-91322, 2006.

[2] H. Termuehlen, *100 years of power plant development*, ASME Press, New York, 2001.

[3] V. K. Der, "Vision 21 – A program for clean energy in the new millennium," *13th U. S. – Korean joint workshop on energy and environment*, September 1999.

[4] R. T. Sawyer, *The Modern Gas Turbine*, Prentice-Hall, New York, 1945.

[5] L. E. Bakken, K. Jordal, E. Syverud, and T. Veer, "Centenary of the first gas turbine to net power output: A tribute to AEgidius Elling," *ASME Paper* No. GT2004-53211, 2004.

[6] H. Lukas, "Survey of alternative gas turbine engine and cycle design," *EPRI Report*, AP-4450, Research Project 2620-2, February 1986.

[7] E. Jeffs, "Franz Stolze: gas turbine engineer ahead of his time," Gas Turbine World, May-June 1986.

[8] D. Eckardt and P. Rufli, "Advanced gas turbine technology - ABB/BBC historical firsts," *ASME Paper* No. 2001-GT-0395, 2001.

[9] The world's first industrial gas turbine set at Neuchatel (1939)-An international historic mechanical engineering landmark, *ASME publication*, September 2, 1988.

[10] A. J. Scalzo, R. L. Bannister, M. DeCorso, and G. S. Howard, "Evolution of heavy-duty power generation and industrial combustion turbines in the United States," *ASME Paper* No. 94-GT-488, 1994.

[11] D. E. Brandt, "The design and development of an advanced heavy-duty gas turbine," *J. of Eng. for Gas Turbines and Power*, Vol. 110, pp 243-250, 1988.

[12] D. L. Chase, "Combined Cycle development evolution and future," GE Power Systems, *GER-4206*, April 2001.

[13] H. Termuehlen, "Forty years of combined cycle power plants," *Energy-Tech*, ASME Publication, October 2002.

[14] G. A. Cincotta, "The testing of GE's MS7001F gas turbine," *Diesel & Gas Turbine Worldwide*, January - February 1991.

[15] R. L. Bannister, N. S. Cheruvu, D. A. Little, and G. McQuiggan, "Development requirements for an advanced gas turbine system," *ASME Paper* No. 94-GT-388, 1994.

[16] S. A. Ali and H. A. Webb, "Advanced turbine systems program requirements and an approach to implementation," *ASME Paper* No. 94-GT-411, 1994.

[17] J. C. Corman, "Gas turbine power generation evolutionary advances for the future," *ASME Paper* No. 95-GT-271, 1995.

[18] A. W. Layne, "Developing the next generation gas turbine systems – A national partnership," *ASME Paper* No. 2000-GT-176, June 2000.

[19] I. S. Diakunchak, G. R. Gaul, G. McQuiggan, and L. R. Southall, "Siemens Westinghouse advanced turbine systems program final summary," *ASME Paper* No. GT-2002-30654, 2002.

[20] R. D. Brdar and R. M. Jones, "GE IGCC technology and experience with advanced gas turbines," GE Power Systems, *GER-4207*, Oct. 2000.

[21] J. H. Horlock, "The evaporative gas turbine (ECT) cycle," *ASME Paper* No. 97-GT-408, 1997.

[22] A. Lazzaretto, and F. Segato, "Thermodynamic optimization of the HAT cycle plant structure-Part I: Optimization of the Basic plant configuration," *J. of Eng. for Gas Turbines and Power*, Vol. 123, pp 1-7, 2001.

[23] M. Nakhamkin, and J. R. Schiebel, "CHAT technology: an alternative approach to achieve advanced turbine systems efficiencies with present combustion turbine technology," *ASME Paper* No. 98-GT-143, 1998.

[24] S. E. Veyo, S. D. Vora, K. P. Litzinger, and W. L. Lundberg., "Status of pressurized SOFC/Gas turbine development at Siemens Westinghouse," *ASME Paper* No. GT-2002-30670, 2002.

[25] A. D. Rao, G. S. Samuelsen, F. L. Robson, and R. A. Geisbrecht, "Power plant system configurations for the 21st century," *ASME Paper* 2002-GT-30671, 2002.

[26] P. Schwarzbozl, R. Buck, C. Sugarmen, A. Ring, M. J. M. Crespo, P. Altwegg, and J. Enrile, "Solar gas turbine systems: Design, cost and perspectives," *Solar Energy*, 2005 (Elsevier publication).

[27] S. Consonni and E. Macchi, "Gas turbine cycles performance evaluation", ASME *COGEN_TURBO Meeting*, Montreaux, Switzerland, September 1998.

[28] M. A. El-Masri, " On Thermodynamics of Gas-Turbine Cycles: Part 2 - A Model for Expansion in Cooled Turbines" *J. of Eng. for Gas Turbines and Power,* Vol. 108, pp 151-159, 1986.

[29] R. Bettocchi, G. Cantore, G. Negri di Montenegro, and D. Moro, "Optimum performance evaluation in multistage intercooled compression gas turbine", *ASME COGEN - TURBO IGTI* Vol. 6, pp. 235-242, 1991.

[30] I. G. Rice, "Thermodynamic evaluation of gas turbine cogeneration cycles: part II. Complex cycle analysis", *J. of Eng. for Gas Turbines and Power*, Vol. 109, 1987.

[31] M. Bianchi, G. Negri di Montenegro and A. Peretto, "Thermo-Economic Optimization of a Cogeneration Plant With Below Ambient Pressure Discharge Gas Turbine", *ASME Paper* No. 2001-GT-209, 2001.

[32] R. K. Bhargava and C. B. Meher-Homji, "The Historical evolution of gas turbine technology for land, sea, and air," *A poster developed to commemorate 50th anniversary of IGTI*, 2005.

[33] H. Lukas, "Cycles Revisited," *IGTI News Letter*, August 1994.

[34] H. Abdallah, and S. Harvey, "Thermodynamic analysis of chemically recuperated gas turbines," *Int. J. Therm. Sci.* Vol. 40, pp372-384, 2001 (Elsevier publication).

[35] P. A. Dellenback, "Improved gas turbine efficiency through alternative regenerator configuration," *J. of Eng. for Gas Turbines and Power*, Vol. 124, pp 441-446, 2002.

[36] K. F. Kesser, J. F. Hoffman, and J. W. Baughn, "Analysis of a basic chemically recuperated gas turbine power plant," *J. of Eng. for Gas Turbines and Power*, Vol. 116, pp 277-284, 1994.

[37] K. K. Botros, M. J. de Boer, and H. G. Fletcher, "Thermodynamics, environmental economic assessment of CRGT for exhaust heat recovery in remote compressor station applications," *ASME Paper* No 97-GT-510, 1997.

[38] R. K. Bhargava, "Gas turbine inlet air cooling technologies," Invited Lecture at National Institute of Technology, Jamshedpur, India, January 2007.

[39] M. J. Reale, "New High Efficiency Simple Cycle Gas Turbine-GE's LMS100™", GE Power Systems *GER - 4222A*, 2004.

[40] C. Jones and J. A. Jacobs III, "Economical and technical considerations for combined-cycle performance - enhancement options," GE Power Systems, GER 4200, October 2000.

[41] S. B. Shepard, T. L. Bowen, and J. M. Chiprich, "Design and development of the WR-21 intercooled recuperated (ICR) marine gas turbine," *ASME Paper* No. 94-GT-79, 1994.

[42] M. A. El-Masri, "A modified high efficiency recuperated gas turbine cycle," *J. of Eng. for Gas Turbines and Power*, Vol. 110, pp 233-242, 1988.

[43] M. A. El-Masri, "Thermodynamics and performance projections for intercooled/reheat/recuperated gas turbine systems," *ASME Paper* No. 87-GT-108.

[44] Dresser-Rand Periodical Publication, "Compressed air energy storage: A technology whose time has arrived," *Insights*, Vol. 3, No. 2, 2000.

[45] E. D. Larson and R. H. Williams, "Steam injected gas turbines", *J. of Eng. for Gas Turbines and Power*, 1997, Vol. 109, pp. 55-63.

[46] D. Y. Cheng, and A. L. C. Nelson, "The chronological development of the Cheng cycle steam injected gas turbine during the past 25 years," *ASME Paper* No. GT-2002-30119, 2002.

[47] J. Tuzson, "Status of steam-injected gas turbines," *J. of Eng. for Gas Turbines and Power*, Vol. 114, pp 682-686, 1992.

APPENDIX A

Table A1 Early development of successful industrial gas turbine systems

System No.	Year of Operation	Cycle	Power (kW)	TIT ˚F (℃)	PR	Efficiency LHV (%)	Manufacturer Developer	Installation Location	Comments
1	1903	Brayton	8.1	752 (400)	N/A	N/A	AEgidius Elling	Oslo, Norway	Mixture of gas and steam at turbine inlet
2	1905	REC	150	752 (400)	2.5	N/A	Franz Stolz	Weissensee Power Station, Germany	First complete axial design, Silo combustor
3	1926	ICR	N/A	900 (482)	9.2	N/A	Brown Boveri Cie (BBC)	Leuna Synthetic fuel plant, Germany	Process energy recovery
4	1939	Brayton	4000	1022 (550)	4.4	17.4	BBC	Neuchatel, Switzerland	First central power station, Historical Landmark
5	1944	REC	2,611	1500 (816)	4.0	17.7 (HHV)	Allis-Chalmers	US Navy Test facility	Separately fired power turbine (parallel combustors)
6	1945	ICR	10,000	1070 (577)	12.0	23.0	BBC	Filaret Station, Romania	
7	1946	ICRHR	27,000	1202 (650)	8.1	34.0	BBC	AG Beznau Station,	Complex cycle, twin shaft, two intercooler
8	1949	Brayton	3,500	1400 (760)	6.0	N/A	GE	Oklahoma Gas & Electric, Belle Isle Station, USA	Evaporative cooling and heat recovery to heat feedwater, ASME Landmark
9	1950	ICR	5,000	1500 (815)	9.0	28.0	GE	Bangor Hydroelectric, Maine, USA	Two recuperators and intercoolers
10	1950	Semi-closed ICRHR,	20,000	1200 (649)	20.0	32.0 (HHV)	Sulzer Brothers	Weinfelden Station, Switzerland	

Table A2 Performance of industrial gas turbine systems 1950-present

System No.	Year of Operation	Cycle	Power (kW)	TIT ˚F (℃)	PR	Efficiency LHV (%)	Manufacturer Developer	Installation Location	Comments
1	1951	Closed, IC	2,000	1220 (660)	3.6	24.0 (HHV)	John Brown & Co.	Atlnabreac, Scotland	Fuel - Peat and coal slurry
2	1979	REC	5,130	(910)	7.6	31.4	MAN Turbo, THM 1203R		
3	1986	Cheng	49,000	2200 (1205)	27.6	41.0	GE LM5000	Simpson's Paper Co.	
4	1990	CC	214,000	2300 (1260)	13.5	45.2	GE 7F	Chesterfield No. 7 power station, USA	First Installation
5	1992	Brayton	43,000	2300 (1260)	29.0	40.0	GE LM6000	Ottawa, Canada	First industrial GT with 40% eff. Aeroderivative
6	1993	CC	160,000	2300 (1260)	14.6	51.7	Westinghouse 501F	Lauderdale plant of FP&L, USA	First Installation
7	1994	Brayton	152,700	2350 (1288)	16.0	36.1	Siemens V84.3	Portland Station Pennsylvania, USA	First Installation
8	1994	RH	179,000	2300 (1260)	32.0	37.5	Alstom GT24	Poryong, South Korea	
9	1997	ICR	25,252	1800 (982)	16.2	41.2	Mitsubishi-Rolls-Royce WR-21	Royal Navy's Type 45 D class destroyer	First Installation
10	1997	CC	330,000	2732 (1500)	20.0	58.0	MHI M501G	Takasago, Japan	Steam-cooled combustor
11	2003	CC	480,000	2606 (1430)	23.0	60.0	GE 9H	Baglan Bay, UK	First gas turbine to achieve 60% overall cycle efficiency and steam cooled blades
12	2003	Brayton	4,600	(1163)	9.9	38.5	Solar, Mercury-50		Solar's highest efficiency gas turbine
13	2006	IC	100,000	2550 (1380)	42.0	46.0	GE LMS-100	Groton, SD, USA	Highest Efficiency industrial gas turbine